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MTI-65TR5-IV

# EXAMPLE APPLICATION OF GAS BEARINGS TO SHIPBOARD MACHINERY

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J. W. Bjerklie
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**TECHNICAL REPORT**

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## INTRODUCTION

This part of the report illustrates the application of gas bearings to an actual machine design. By showing all the effects that must be checked, and by using the criteria and curves presented in MTI-65TR5-II, Volumes 1 and 2, it is hoped the reader will be able to verify or evaluate the practicability of particular gas bearing designs for particular machines.

It is concluded in MTI-65TR5-III that Forced-Draft-Blowers represent a suitable candidate for use with gas bearings. It is decided to use a forced-draft-blower as the example for practical application of gas bearing design since there is considerable Navy interest in such an advanced design. It was further decided to design a unit superior to those presently in use. This would have the further advantage of becoming an even more likely candidate for the use of gas bearings. The new design exploits advantages to be gained through the use of process fluid lubrication, and great improvement in reliability and maintainability is predicted.

The advantages to be expected through the proper use of PFL (process fluid lubrication) are substantial and direct. The most unreliable components in machinery are very often its bearings and seals. Seals are usually required to separate two process fluids and to keep the lubricating oil from the process fluid and vice versa. Using the process fluid as the lubricant greatly simplifies the demands on the seals — in some cases eliminating the requirement for a seal altogether. Journal bearings very often fail because of lubricant supply failure or because of lubricant contamination. The use of process fluid lubrication eliminates the conventional lubrication system with its inherent unreliability and also eliminates any concern for contamination of the lubricant by the process fluid — for example, contamination of the lubricating oil by condensate in a steam turbine. Now, if a gas is available as the process fluid, additional benefits can be gained through design capitalization on the characteristics of gas bearings. Gas bearings can be designed to be insensitive to high speeds and high temperatures. Thus, the temperature and speed limitations of conventional bearings on design can be greatly diminished. In summary, PFL

can lead directly to a major increase in reliability, decrease in maintenance and, where gas lubrication is possible, a major reduction in size and weight.

In MTI-65TR5 - III, an evaluation seeking the best machinery candidates for gas bearings or PFL is reported. Several typical kinds of naval turbomachinery were considered; but, in some cases a residual problem existed to prevent full realization of the potential benefits. For example, a main propulsion turbine driving a reduction gear could be designed for use with either steam or water bearings. However, the gears would need a separate lubrication system so that the overall gain with steam or water lubrication of the turbine would be small. In some of the other applications, major developments are required in such areas as high-speed generators, high-head pumps and low-noise turbomachinery — in addition to process fluid lubrication. However, in the case of a forced draft blower, full realization of the potential of process fluid lubrication appears to be possible. The blower is steam driven and direct-connected so that a steam bearing system can completely replace the current lubrication system.

This bearing application study relates the bearing design to the curves of MTI-65TR5 - II and the conclusions of MTI-65TR5 - I and III. Wherever required, the discussion and design curves of MTI-65TR5 - II are specifically referred to so the reader can follow the complete design process. In addition, the important system features to be checked to determine the adequacy of the design are cited. Thus, stress, temperature, steam conditioning, and rotor response are checked. Since the major topic of this report is bearing design, these other aspects are not emphasized. It is sufficient that they be called to the attention of the engineer since the methods of calculation are standard.

The Navy specifications for FDB's (forced draft blowers) are presented in MIL F-19602A (SHIPS), dated 1 August 1963. For an advanced design, however, some parts of the specs would not apply at this time. Consequently, the new FDB was designed to modified specs that aimed only at achieving several salient features. Detailed specifications for materials, testing, shipping, etc., could then be by-passed for this example. The modified specification is quite simple and is given in the following section.

# PRELIMINARY SPECIFICATION FOR ADVANCED DESIGN FORCED DRAFT BLOWER

## 1.0 SCOPE

1.1 This specification covers steam turbine driven, forced draft, vane-axial fans for supplying combustion air to the main steam boiler on steam driven Naval ships.

## 2.0 DESIGN

2.1 General design - The principle of reliability is paramount and no compromise of this principle shall be made with any other basic requirements of design. It is the intent of this specification to obtain forced draft blowers of such design that they will operate over a long period of years with a minimum of servicing. Where wear, erosion or corrosion is unavoidable, the parts subjected to such detrimental effects shall be of the best material available for the purpose in order to reduce those effects to a minimum. The design and construction of all blowers shall be the most compact possible, consistent with the following requirements.

- (a) Reliability
- (b) Accessibility for repair
- (c) Economy
- (d) Satisfactory operation when inclined as follows:
  - (1) Up to 15 degrees in any direction (permanently inclined)
  - (2) With the ship rolling up to 45 degrees from the vertical to any side
  - (3) With the ship pitching 10 degrees up and down from the normal horizontal plane
- (e) Minimum space and weight

2.2 The following design conditions shall be assumed:

- (a) Intake air dry bulb temperature - 68 degrees Fahrenheit (F)
- (b) Intake air pressure - 29.92 inches mercury (Hg)
- (c) Relative humidity - 50 percent
- (d) Ambient temperature - 140 F

2.3 Each forced draft blower shall be a self-contained unit consisting of a casing with guide vanes, propellers, and a drive turbine with integral lubrication system. The propellers and turbine wheels shall be mounted on a common shaft.

## 3.0 MOUNTING

3.1 Blowers shall be designed for horizontal or vertical mounting.

3.2 Each horizontal blower with all appurtenances shall be mounted on a common bedplate.

## 4.0 PARALLEL OPERATION

4.1 All blowers shall be of a design to insure satisfactory operation of two or more units of the same type in parallel. Each blower shall be capable of being brought from idling speed into stable parallel operation with one or more blowers already operating at their rated capacity and 45 percent of their rated static discharge pressure without the speed of the oncoming blower exceeding that of the operating units, and once in parallel, shall remain in parallel under a minimum speed difference of 300 revolutions per minute (r.p.m.) between units.

4.2 Under required methods of operation on Naval ships, blowers may be required to operate together at similar reduced speeds (less than 45 percent of rated speed) with one or more, but not all, operating in an unstable (stall) region over extended periods of time.

4.3 All blowers may be operated at standby, that is at minimum speeds, capacities, and pressure for indeterminate periods of time but shall be capable of being instantly speeded up to maximum operations; and when so operating shall be capable of being instantly slowed down to minimum requirements; as when maneuvering. Minimum speed is defined as 15 percent of rated speed.

4.4. Unless otherwise specified, it shall be assumed that blowers will take suction from a plenum chamber and discharge air to the boilers through

ducts. There may be several blowers in one plenum, but generally there will be an individual discharge duct for each blower.

## 5.0 SHOCK AND VIBRATION

5.1 The design of the complete forced draft blower shall be such that it is capable of sustaining shock and vibration loads of intensity and frequency normally encountered in service.

5.2 Shock mounts shall not be used.

## 6.0 STEAM TURBINES

6.1 Steam conditions shall be greater than 350 PSIA and greater than 1000 F at the design point, but may be 200 PSIA, saturated, for part of the duty cycle.

6.2 The combined blower-turbine efficiency rated blower capacity and static pressure shall not be less than that tabulated in Table IV-1.

Table IV-1 - Minimum combined efficiency at blower rating - percent

| Static pressure at blower rating - inches $H_2O$ | Capacity at blower rating - c.f.m. | | |
|---|---|---|---|
| | 20,000 | 30,000 | 40,000 |
| 120 | | 41 | 42 |
| 110 | | 38 | 39 |
| 100 | 31 | 35 | 36 |
| 80 | 29 | 32.5 | 34 |
| 60 | | 30.5 | 31.5 |
| 40 | 25 | 26.5 | 27.5 |

Note 1. The combined blower-turbine efficiency is defined as

$$\frac{\text{Air h.p. x Theoretical water rate}}{\text{Actual steam consumption}}$$

"Air h.p." is defined as $\frac{\text{CFM x S.P.}}{6356}$ where CFM refers to standard air. "Theoretical water rate" is defined as the theoretical pounds of steam required per horsepower-hour under the design steam conditions.

Note 2. Minimum efficiencies in Table IV-1 refer to the blower rating point only, and do not cover any other point of blower operation.

6.3 Steam leakage from the turbine glands shall not exceed 370 pounds per hour per blower throughout the blower operating range and under the following conditions:

(a) 15 pounds per square inch gauge (psig) turbine exhaust pressure.
(b) 10 inches $H_2O$ vacuum maintained at the turbine gland exhaust connection for labyrinth seals.
(c) 2 inches $H_2O$ vacuum maintained at the turbine gland leak-off connection, for carbon packing glands.
(d) Gland seals worn to twice the design clearance, or at the manufacturer's recommended maximum service clearance; whichever is largest.
(e) No steam leakage from shaft glands to machinery space (as evidenced by positive air intake from space into gland).

6.4 Air entrance into shaft glands and other seals, and exhausted by the gland exhaust or gland leak-off system, shall not exceed 40 pounds per hour per turbine under the conditions specified in 6.3

6.5 The speed control system shall not require any hydraulic tubing or piping leading away from the blower or attachments thereto.

6.6 A speed limiting governor shall be furnished as an integral part of each blower. It shall positively limit turbine speed under any normal or abnormal operating condition (as when system resistance is lost or when blower propeller blades are sheared).

## 7.0 BEARINGS

7.1 Radial and journal bearings shall be steam lubricated utilizing steam from the turbine supply.

## THE DESIGN

The forced draft blower design presented herein is preliminary, with emphasis being placed on it as an example of gas bearing usage. Figure IV-1 shows the layout of the unit.

The major system problems are contamination, noise, steam leakage, poor efficiency, bearings and seals. An additional problem is the difficulty of inspection and accessibility for routine maintenance.

This concept will meet Navy specifications because of the design freedom given by the use of PFL. Several clear-cut advantages over existing conventional equipment are shown:

- Reliability

  Greatly increased because of elimination of a lubricant supply system and the elimination of any contamination problems from condensate leakage.

- Maintenance

  Reduced through machine simplification. Wear is also reduced because of the vibration reduction achieved by shortened shaft length.

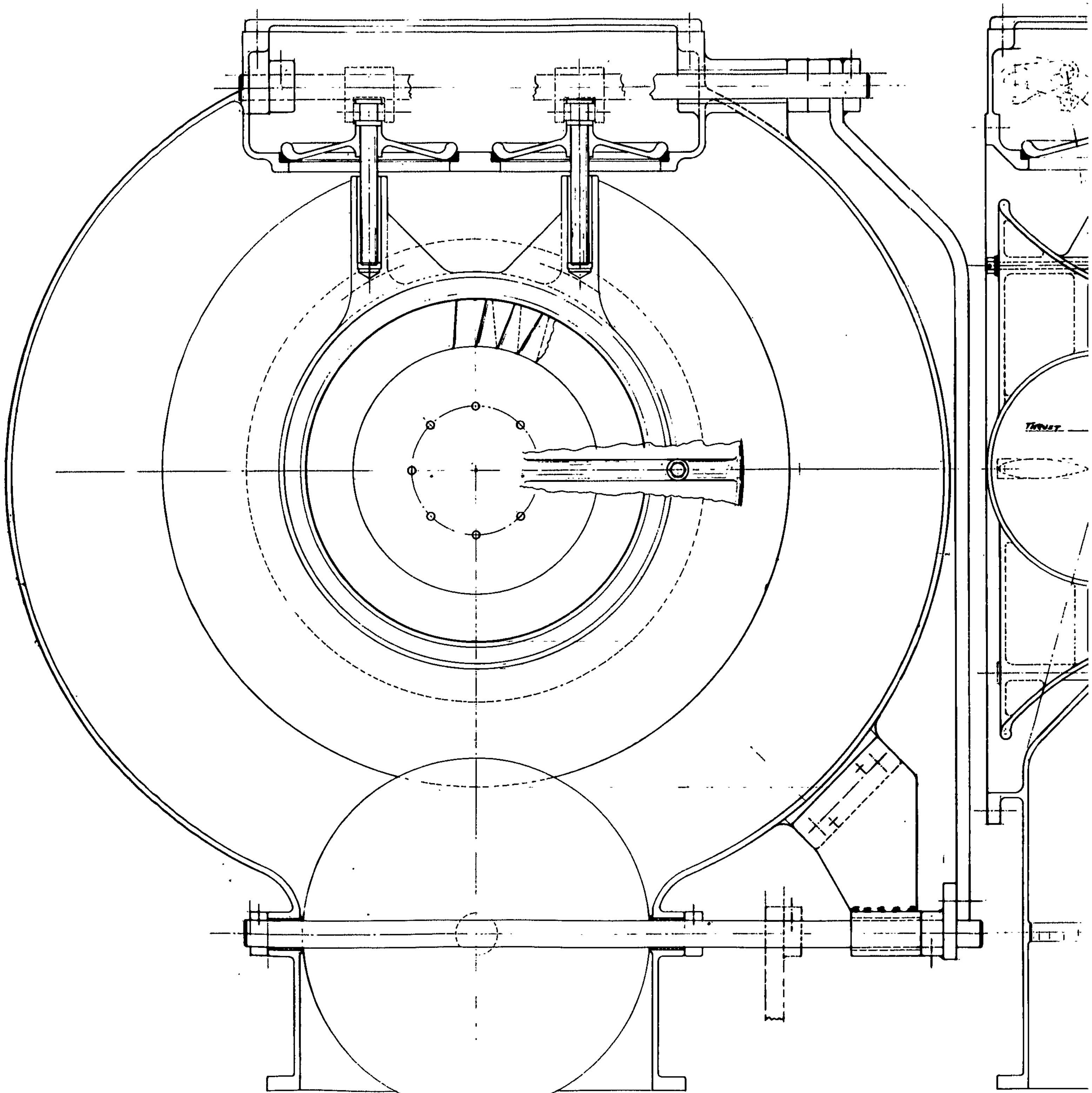
- Size and Weight

  Reduced through compact design and high speed.

- Accessibility

  Greatly improved because of compact design.

A discussion of this design is presented here with detailed consideration of the bearing design.



SECTION A-A

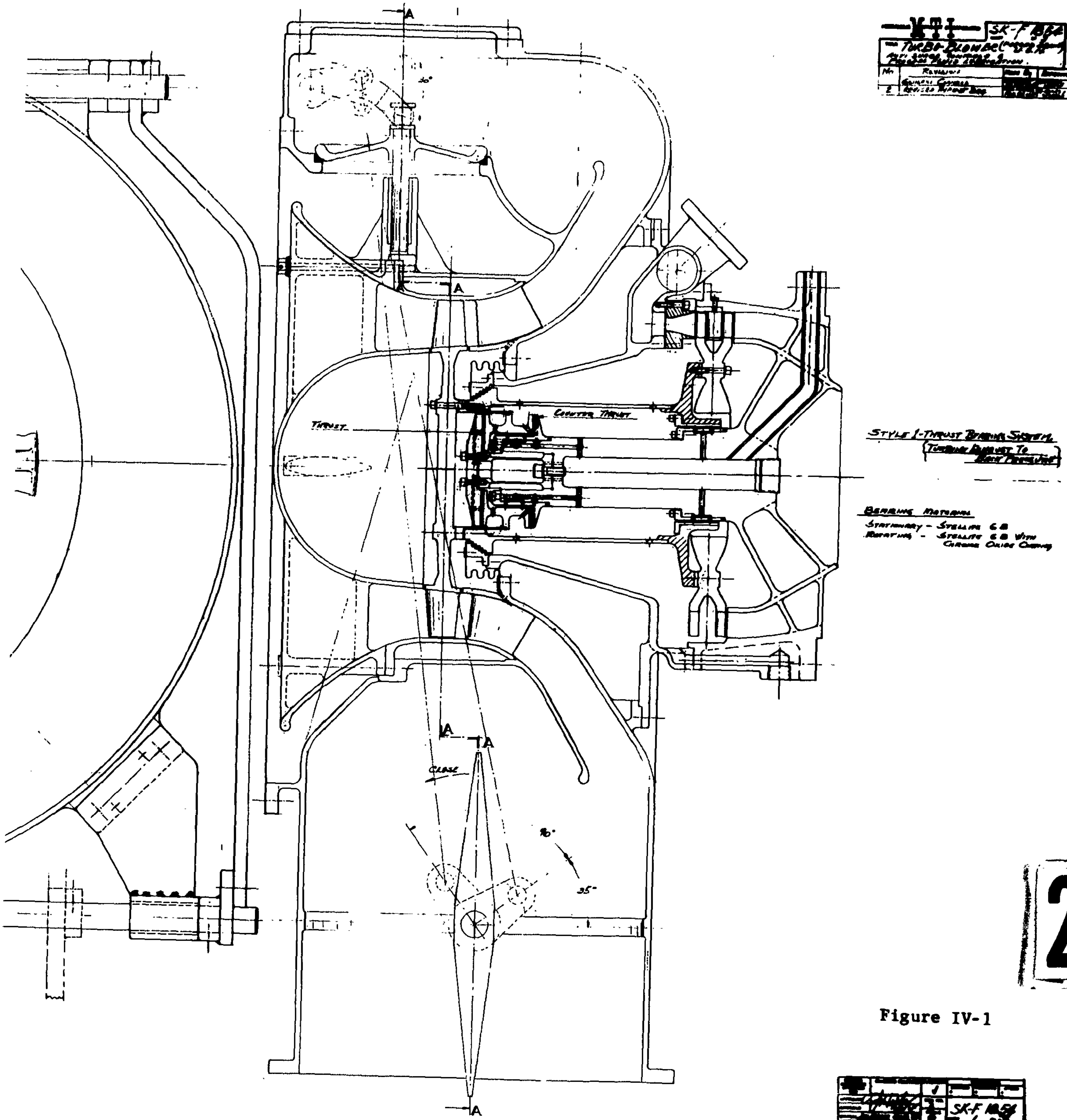

TURBO-BLOWER
THRUST
COUNTER THRUST
STYLE 1-THRUST BEARING SYSTEM
BEARING MATERIAL
STATIONARY - STELLITE 6B
ROTATING - STELLITE 6B WITH CHROME OXIDE COATING
CLOSE
A
35°
2
SK-F

Figure IV-1

## A. Design Philosophy

The forced-draft-blower envisaged here is a high-volume, high-speed fan driven by a steam turbine to force air into the furnace on Naval ships. The basic characteristics expected of the design are to:

- Eliminate use of a third fluid as a lubricant.
- Eliminate the need for auxiliary equipment for lubrication.
- Separate steam from air, and prevent leakage of steam into the surroundings.
- Achieve easy assembly and maintenance.
- Achieve minimum unit size and weight.
- Achieve high reliability while meeting specified performance.

Achievement of these goals is accomplished by the following basic machine design philosophy:

- Use steam as the lubricant, thereby limiting dirt and contamination in the system.
- Use a hollow rotating shaft with internal bearings and the short-coupled blower and turbine wheels.
- Use as high a rotating speed as practical.
- Simplify the design to a minimum of parts without sacrificing performance.

Application of this philosophy has resulted in a conceptual design which can be compared to present Navy blowers as follows:

| | MTI Concept | Present Navy Blower (approximate) |
|---|---|---|
| Blower Diameter | 19-1/2 inches | 36 inches |
| Blower Stages | 1 | 1 |
| Turbine Diameter | 15.7 inches | 36 inches |
| Turbine Stages | 1 | 1 |
| Shaft Length | 17 inches | 72 inches |
| OA Length | 37 inches | 72 inches |
| OA Weight | 850 lbs. | 2000 lbs. |
| Mounting | Vertical or horizontal | Vertical |
| Lubricant | Steam | Oil |

This comparison indicates a definite advantage for the proposed unit. Overall advantages resulting from the concept are:

- A single, basic design can be adapted to widely varying applications.
- It can be mounted in any position (vertically, slanted or horizontal).
- Maintenance is easy and economical because of the simple assembly and few parts.
- Overall reliability is high.
- Size and weight of this unit are low.
- Intermixing of steam and air is virtually eliminated, even though only low-pressure seals are used.
- Overall efficiency is excellent.
- Means are provided to positively prevent overspeed and surge damage.
- Any steam condensate from the bearings is dumped into the turbine exhaust steam.
- The shaft is short and stiff, exhibiting easily-controlled rotor dynamics.
- Thrust loads are easily controlled.
- Ducting is efficient, and machine layout is large enough to keep thermal gradients down even though the unit is compact.
- Non-exotic materials can be used.
- Parasitic losses are low, including turbine and blower losses, and especially low bearing losses.

Basically, the blower draws air in, compresses it, and discharges it into a scroll surrounding the diffuser. The blower is driven by a direct-connected steam turbine by means of a short shaft. The steam flow is delivered from the main boiler at up to 1000 psi and 1000 F. It is then fed to the turbine through two nozzles to maintain thermal symmetry. Overall unit sizing is kept down by accepting higher rotational speeds consistent with "near optimum" component efficiencies and without introduction of excessive stress or noise.

The bearings are mounted inside the short hollow shaft connecting the turbine and blower. They are, in turn, supported on a stationary shaft cantilevered from the main support structure. The bearings are steam-lubricated and consist of two journal bearings, a thrust bearing and a positioner bearing opposing the thrust bearing. Superheated steam is supplied to the bearing system.

Reliability and ease of maintenance are inherent in the proposed unit. With the use of hydrostatic bearings, the only wear is, possibly, erosion and corrosion due to feeding lubricant into the bearing. Compatible materials must be chosen so as to minimize wear under conditions of high-speed rub which may

occur under severe shock. An experimental study of the affects of shock and ship motion was reported under Contract Nonr-4358(00).

Present day bearing materials limit wear during rubbing to values which can be tolerated over any normal blower lifetime. Other components of the machine are not ordinarily subject to wear — although fatigue and creep play a part in determining unit life. The top speed and temperatures associated with the machine are well within the present state-of-the-art for materials life. Consequently, reliability is actually determined by any problems encountered in installation or operation of the unit rather than by such items as wear, erosion, corrosion, or creep and fatigue.

Maintenance of this unit will be easy. The entire rotating assembly can be lifted off its mounting post. Thereafter, every part is available for close scrutiny with very little additional unit disassembly. The absence of an oil lubrication system and of oil-lubricated bearings greatly reduces the number of parts needing regular inspection and reduces the time and expense of disassembly and assembly of the blower. It also eliminates the inspection of auxiliary equipment since it is not required.

The design, as presented, does not necessarily represent a final design approach. For instance, heaters are presently deemed necessary at the bearings. It may be found that these are not really required in an actual operation, but it will not become perfectly clear that this is so until the whole unit is subjected to detailed design. Consequently, the use of these auxiliaries in this design represent preliminary thoughts only, and should not be considered basic to the design. In a test program during development it can be determined whether or not the auxiliaries to be discussed (bearing heater, steam superheaters, electronic speed control) will be needed or are the best possible alternative. Also, it is during test that final selection of clearances and orifice size can be made.

## B. General Design Consideration

The general layout was selected to accomplish separation of steam and air while maintaining a sound mechanical design. A conventional arrangement with a rotating shaft supported by two journal bearings and having the blower and turbines at opposite ends of the shaft immediately poses several problems. Air,

high-pressure steam and low-pressure steam all occur at some point on the shaft — thereby necessitating extensive sealing. By placing the bearings with their high-pressure steam at the inside of the shaft, rather than having them next to either ambient or system air on the outside, steam leakage sources are isolated so that the flow must go to the steam exhaust system. Only low pressure seals are then required between steam and compressed air.

The length of the shaft will be determined by bearing location, stability consideration, fluid ducting and sealing. By placing the bearings inside the shaft, the support function can be accomplished almost independent of the blower and turbine locations. The remaining criteria for shaft length selection then becomes a matter of rotor response, fluid ducting and sealing. Actually, ducting can be adapted for any design — from back-to-back or inside-outside turbine blower wheels to very long shafts. However, very long shafting introduces factors such as low critical speeds, high damping requirements, etc., that make it unattractive for a good design. Hence, it appears desirable to operate with a shaft as short as possible. A limit is reached when ducting becomes expensive, the distance between bearings becomes so short that conical modes of shaft instability are very easily set up and/or leakage between steam and air becomes excessive. The shortest machines, back-to-back or inside-outside blowers and turbines on single wheels, pose the most severe problems with sealing between the air and steam systems. The inside-outside wheel poses the most severe stress picture along with the most severe sealing problem.

For the above reasons it was deemed desirable to keep the shaft length on the order of 18 inches in length. The design reflects this decision.

Figure IV-2 gives the Pressure-Flow relation for a blower. The surge line is a limiting operating condition, and all operation should be restricted to the area at the right of the surge line. Forced-draft blowers may be operated for long periods of time in the area to the left of the surge line — such as during ship maneuvers. The surge condition could lead to machine failure if allowed to persist; especially, it could lead to excessive vibration, noise, blade fatigue and possible bearing failure. This is remedied in this concept by the use of a mechanism linking the discharge shut-off valve to a by-pass valve which allows blower exhaust to recirculate to the inlet. This allows

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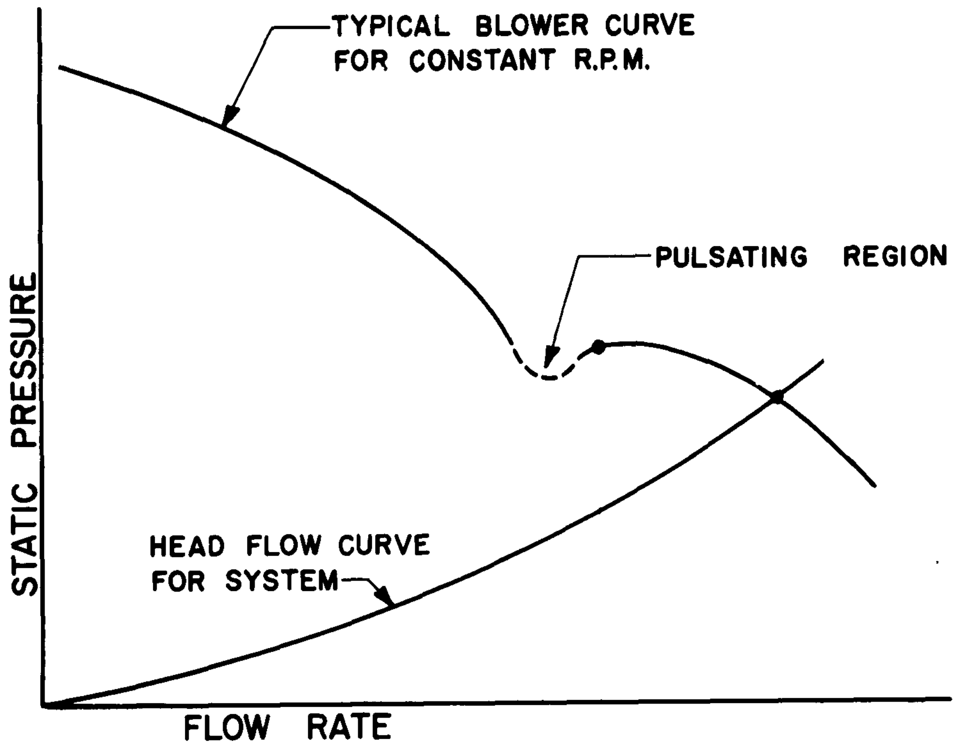


Fig. IV-2 Pressure versus Flow

blower flow to remain high although discharge flow is small. Therefore, surge is prevented. The response of the system to surge conditions needs to be determined before the final anti-surge system can be devised.

The linking could be accomplished by a servo-motor or by mechanical linkage. However, a mechanical linkage of one valve to the other seems to be straightforward and introduces fewer components into the reliability train. The mechanical system is shown in Fig. IV-1.

The forced-draft-blower speed may be controlled by mechanical, pneumatic, electronic, or pure fluid control components. Also, combinations of these control types may be employed. Because of the advanced level of development and demonstrated reliability, a system incorporating electronic speed sensors and steam flow valve control seems to be attractive for this application.

The blower speed control system easily can regulate to within one percent of desired value. This is about 2-1/2 times better than that required by the U. S. Navy specification. Other types of control can also be used to meet the specifications. The unit operates at 12,500 RPM with regulation within 300 RPM being desired.

Steam conditions are variable and run the gamut from very high pressures and superheat temperatures to low pressures and saturated steam conditions. This is dependent on the type of installation and specific operating conditions for a given installation. With change in design conditions, it is necessary that a means be provided to allow easy adjustment within the unit. In any case, each installation will have a design condition and must operate "off design" when the ship's steam supply varies. The variation in available steam supply considered here is given below.

| | |
|---|---|
| Pressure | 600-1000 psi |
| Temperature | 800-1000 F |

Different nozzle blocks can be inserted into the turbine inlet housing for different system steam characteristics. Nozzle block changes alone provide a gross adjustment. A proper set of turbine blades together with a new nozzle block provides complete adjustment of a single basic machine to various applications.

The blower can be a precision casting — depending on the final design. The inlet throat piece, with vaned air guidance and nosepiece, is removable for replacement if the air flow conditions are changed from one application to the next to the extent that a new impeller design is required. The diffuser guide vane is also replaceable to match a new impeller.

The blower housing is centered on the turbine housing through the four, radial-splined legs. These splines allow the turbine housing to thermally expand without concentricity misalignment.

The steam turbine requires partial-arc admission. The design incorporates two nozzle blocks 180 degrees apart to reduce force moments on the shaft and to induce more uniform heating of the structures. The wheel disk is attached to the hollow shaft by a radial-splined connection which transmits torque but allows the disk to thermally and mechanically expand without affecting the shaft. The turbine incorporates two rows of blades with stationary guide vanes between these velocity stages. The guide vanes are bolted into place at each nozzle block, and do not extend all the way around the turbine.

The turbine inlet manifold is shown with one steam inlet at the top. It extends halfway around the circumference to the other nozzle block area. The nozzle blocks are bolted to the manifold and are replaceable.

The turbine exhaust hood is the backbone for the entire turbine assembly. The large flange may be bolted to a steam exhaust piping system. To this structure there is attached a cantilevered post on which the hollow shaft rides. The lubricant supply is piped through the cantilevered post to the bearings.

The bearing type was selected keeping in mind the available gases, the required speed range, and imposed shock and vibration. Low-pressure air and high-pressure steam are available to the bearing. Either will serve as a lubricating gas. The imposed shock, vibration and ship motion are moderate and either hydrodynamic or hydrostatic bearings will satisfy the requirements. The unit must be operated at speeds corresponding to idling up to 12,500 RPM. Idling speed is defined as 15 percent of rated speed, or 1875 RPM.

The speed range is sufficiently wide to eliminate hydrodynamic bearings from further consideration. It would be possible to design them, but the speed variation is too great for a good operating margin at all times. The choice of hydrostatic bearings eliminates air as a lubricant since the highest pressure available at the system is only slightly over ambient, so the bearings would have to be huge. The use of steam is then mandatory.

With the bearing location selected for this application, the steam and condensate exhausting from the bearings is contained within the tubular shaft and exposed to the turbine exhaust. Therefore, this design easily can isolate the lubricating fluid from the ambient air.

The bearing position allows the critical speed to be quite high. With present knowledge, it is required to operate gas bearing machines below the first bending critical speed — although there is no problem in operating above the rigid body criticals. The first bending critical of the proposed design is on the order of 21,500 rpm.

In comparing this general design with existing forced-draft-blowers, it is immediately apparent that the shaft of this machine is much shorter, the lubrication system considerably simpler, and the entire unit is quite compact.

Merely from consideration of the number of parts, the proposed unit poses a great step forward in maintenance ease.

In comparing this unit to other possibly more compact designs — such as a blower and turbine on the same wheel — it is readily apparent that the proposed unit exhibits much less probable leakage, much less severe stressing, greater shaft stability, and gives the bearings every opportunity to run successfully.

## C. Turbine and Blower Design

The aerodynamic design of the blower reflects the latest developments in technology and uses them to best advantage. This is particularly true because the head requirements approach the operating spectrum of axial compressors. Two representative fan designs are compared below.

The maximum conditions for the blower in the Navy Specification (MIL-F-18602A) are 30,000 SCFM and 40,000 SCFM at 120 inches of water static pressure delivery. A combined blower-turbine efficiency of 42 percent is required. The most stringent design condition is at 30,000 SCFM since this will be the most critical low-speed, high-head combination. The isentropic power at this condition is 532 HP. Using the specification requirements, the blower-turbine isentropic efficiency can be as low as 38.6 percent. Normally this value would be easy to accomplish (65 percent or more could be expected from a normal compressor-turbine combination), but there are important considerations which tend to indicate that the design efficiency should be lower than optimum:

1. A better turbine can be designed at a steam supply pressure lower that specified. By the specification, however, the energy loss in inlet throttling is to be debited against the turbine efficiency. Therefore, the overall performance should be optimized, possibly to the deficit of the blower.
2. Space restrictions on the discharge ducting cause severe problems of efficient diffusion of the fan outlet velocity head. For a fan of few stages, this head will be an appreciable proportion of the total work input.
3. The same head and efficiency is required at both 30,000 SCFM and 40,000 SCFM. Normal fan characteristics show a line of peak efficiency to be at increasing head rise with increasing speed (flow is a function of speed), and the efficiency deteriorates away from this optimum line. Since the required operating line does not follow the optimum line, the design maximum efficiency must, therefore, be set high to allow for the deterioration at off-optimum conditions.
4. The fan must operate with a positive head with acceptable stability at 45 percent speed or less and at zero net flow. It must then be capable of acceleration to normal load conditions, i.e., of progressing from complete stall to normal operation. The blading must, therefore, be capable of mechanically withstanding stall conditions or some means must be found to keep the machine out of stall, e.g., a by-pass system to recirculate air back to inlet. The blading also should be set to give the maximum possible flow range to stall, but if this is done for the 30,000 SCFM condition, then the efficiency deterioration to 40,000 SCFM will be extreme.

## Fan designs

The 30,000 SCFM condition is chosen for design purposes. A full optimization study is required to locate the fan and turbine characteristics relative to this point. For the purposes of this work, the blading will be set to the optimum angles at this condition. The design is made for the following conditions:

| | |
|---|---|
| Flow | = 30,000 SCFM at 68 F; 14.7 pounds/inches$^2$ |
| | = 38.15 pounds/second |
| Pressure rise | = 120 inches $H_2O$ static delivery |
| | = 1.30 pressure ratio; static/total |
| Isentropic temperature rise | = 41 F |

A two-stage fan is a near-future target requirement and this was taken as a design objective. Aerodynamically, a single-stage subsonic fan of high loading is also feasible. However, the ultimate in this direction for the required pressure ratio is a transonic machine, and although this type could undoubtedly have some difficulty in meeting the flow head range specified with a reasonably small variation in efficiency, it serves to define a limit design. A fan temperature rise of 60 F, i.e., a nominal efficiency of 68.5 percent, makes a good design point that can be reached easily by either the subsonic or transonic blower in this flow regime. It is expected that the actual efficiency across the blading will be considerably in excess of this for both designs. The 68.5 percent figure will assure the desired overall efficiency defined in the specification.

With two particular reservations, the transonic design is more attractive than the two-stage subsonic design because of smaller size (lower mass on bearings and faster acceleration), shorter length (much higher critical speed), higher speed (better turbine design conditions) and lower blade and vane loadings (better shape and more efficient characteristics over the blading).

The two reservations are the capability of efficient diffusion of the leaving head, which is a critical problem, and whether the overall characteristics are the most suitable to meet the specified range of head flow combinations with

minimum variation in efficiency. This problem would have to be subjected to further analysis — beyond the scope of this study.

## Turbine

The operating requirements of the forced-draft-blower indicate that during normal operation the fan will require 800 HP. The demand for "off-design" flow can be designed for any requirement. For this reason the turbine is sized on the basis of having available steam at 600 psi and 800 F, throttling down to a lower pressure to provide normal working power, and using full pressure for maximum blower flow rate operation. The steam pressure before the nozzles for the design case was calculated to be 350 psi, which with an exhaust pressure of 15 psig gives an isentropic enthalpy drop of 252 BTU/lb. The corresponding theoretical spouting velocity is 3560 fps, which requires some form of multi-staging to avoid excessive blade speeds. The velocity compound turbine permits the utilization of this enthalpy drop in the smallest number of stages with a mean blade speed of 802 fps. Since no sealing is required in the interstage stator, the two stages can be supported conveniently on a single wheel, thereby obtaining a short overhang and aiding in overall compactness, good rotor dynamic characteristics, and light weight. A hydraulic efficiency of 61.5 percent was confirmed by detailed calculation. The steam flow rate is calculated to be 3.66 pounds per second.

Other turbine types can be used and, in fact, other turbine types can be about as efficient and help balance thrust load better than this one. Consequently, a more thorough study could be undertaken to select a turbine type. In any case, the given figures represent an achievable goal — if not an optimum goal.

## Overall efficiency

At the design condition, the overall efficiency as defined in the specification and for the design condition with either blower and with the velocity compounded impulse turbine is

$$\eta = \frac{\dfrac{\text{SCFM} \times \Delta P_{\text{static}},\ \text{in.}H_2O}{6356} \times \dfrac{2545}{\Delta H_{th}}}{\text{Actual Steam Consumption}}$$

$$= \frac{\dfrac{30000 \times 120}{6356} \times \dfrac{2545}{(262)}}{(3.66 \times 3600)} = .418$$

$= 42$ percent

This is above the specified value of 41 percent for the design condition when the available margin in the first assumptions is considered. Probable peak overall efficiency will approach 46 percent. The good efficiency of this unit should greatly assist start-up operations even if the steam supplied to the turbine is quite far from the design conditions.

## SPECIAL CONSIDERATION

The selection of hydrostatic bearings using steam as a lubricant immediately poses two areas of concern. First, how will the bearing materials tolerate steam; second, what steam conditioning is necessary?

### Erosion and Corrosion

Tests have been carried out elsewhere with steam (Refs. 1, 2, 3 and 4). Wire drawing tests at temperatures of 430 F (saturated), 272 F (saturated), and 500 F (saturated), were carried out over periods of 1000 hrs., 1930 hrs., and 50 days respectively. The results were that Stellite 6, 347 stainless, and some of the 400 series stainless steels were most resistant to erosion and corrosion.

### Steam Conditioning

The steam can be delivered to the blower at 200 psi (saturated) under some circumstances. As discussed in the following Bearing Design section, however, the supply steam must be superheated by at least 100 F at all times. This requires that a steam-conditioning heat exchanger be installed at the blower. In addition, the bearings must always be hot enough to maintain a superheat of about 10-15 F at all times. The blower steam supply pressure is to be reduced by 350 psi as discussed under the previous Turbine Design section.

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1. "Haynes Wear-Resistant Alloys," Copyright, 1959, by Union Carbide Corp.
2. Lichtman, J. Z., Kallas, D. H., Chatten, C. K., and Cochran, Jr., E. P., "Cavitation Erosion of Structural Materials and Coatings," October, 1961 issue of Corrosion Engineering.
3. Letter by Materials and Processes Lab., General Electric Co., to S. F. Murray, dated September 28, 1960.
4. Cataldi, H. A., Cheng, C. F., Musick, U. S., "Investigation of Erosion and Corrosion of Turbine Material in Wet Oxyginated Steam, Trans. ASME Vol. 80, 1958, p. 1465.

This requires that electrical heaters be installed in the blower bearings to maintain temperatures above 450 F. The bearings must be preheated before starting. Results of tests conducted under Contract Nonr-3731(00) substantiate these needs.

The bearing heaters can be easily controlled with permanently-installed thermal controls. Supplying electricity to the blower can be accomplished simply under either sea or dock-side conditions.

The steam-conditioning heat exchanger for bearing steam is also most simply accommodated with electrically heated surfaces. The system would be thermally controlled by the outlet temperature which would be 500 F. A simple, one-pass system using heated rods mounted crosswise in a small duct or the order of 6 in. x 6 in. cross section by 4 in. long will suffice. The rods could be as little as 1/2 inch in diameter mounted on one inch centers. Their temperature need not exceed 1000 F. A higher temperature system (say, 1500 F) would need similar rods and geometry. The heat exchanger would then be on the order of 4 in. x 4 in. x 4 in. Obviously, there is considerable latitude in design of this conditioning heat exchanger, but the main point is that it is small, compact, and can be completely trouble-free by selecting a low enough surface temperature. The steam heater would be about a 12 kw system.

## BEARING DESIGN

As stated before, the selected bearings are hydrostatic with steam lubrication. Steam in the superheat region behaves as any other gas in following the "perfect gas law." It has a viscosity of the same order as most gases, the specific heat ratio is not far removed from that of air and other common gases.

However, the steam pressure within the bearing must be maintained at a level that is higher than the saturation line in order to avoid two-phase flow. This does not present any great problem throughout most of the bearing film since the gas expansion in the film is isothermal. Within the region immediately surrounding the feeder hole entrance, there exists turbulence and possible shock which will lower the pressure. Normally, in a gas-lubricated bearing, the pressure beyond the entrance will recover and can produce a satisfactory pressure profile sufficient to carry design load. In a steam-lubricated bearing, should this pressure drop occur and the pressure fall below the saturation line, a change of phase on the steam will occur from which the pressure cannot recover.

Methods of direct calculation of the entrance-effect pressure drop are not presently known. The calculations performed here will show an approximate method which is considered conservative. This method assumes an isentropic expansion in which pressure drop will be as great as the kinetic energy of the gas.

One other factor used to prevent the formation of condensate in the bearing is the lack of orifices in the supply ports. This will assure that the pressure drop will occur only at the entrance region of the film and not downstream of an orifice. It also assures that the bearing will be inherently compensated which aids in the reduction of pneumatic instability. These aspects are explained in Part II (MTI-65TR5 - II Vol. 2).

A temperature distribution calculation was made on a simplified model using conservative estimates with respect to heat transfer coefficients and source

and sink temperatures. The bearing temperatures are sufficiently high under normal conditions without heaters to prevent condensation of the steam under normal operating conditions.

A stress distribution calculation was made in which the maximum stress in the compressor whell is to be equal to or less than 40,000 psi and in the turbine wheel to be equal to or less than 60,000 psi. The weights obtained are the ones used in this present calculation.

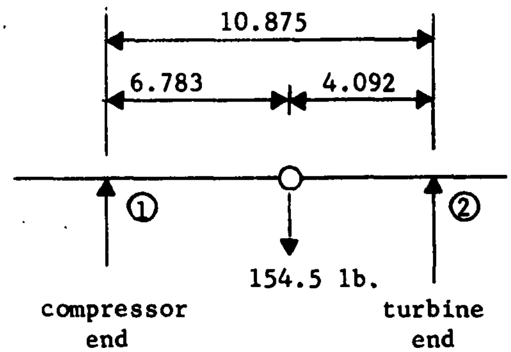
The journal bearings must be able to support the blower rotor which has the following specifications:

| | |
|---|---|
| Rotor Weight, lbs. | 154.5 |
| Journal Bearing Span, in. | 10.875 |
| Location of C.G. (with respect to the compressor end journal bearing center line) | 6.783 |
| Proposed Journal Bearing Diameter, D. In. | |
| Compressor End | 4.0 |
| Turbine End | 5.375 |
| Proposed Journal Bearing Length, L. In. | |
| Compressor End | 2.0 |
| Turbine End | 2.6875 |
| Operating Speed, N. rpm | |
| Design | 12500. |
| Off Design, (15 percent design) | 1875. |
| Gas Supply Pressure, $P_s$, psia | |
| Design | 350 |
| Off Design | 200 |
| Ambient Pressure, $P_a$, psia | 30 |
| Gas Temperature, T, $^oF$ | |
| at Design, $P_s$ = 350 psia | 800 |
| at Off Design, $P_s$ = 200 psia | 382 |
| Gas Constant for Steam, R, in.$^2$ (sec.$^2$ $^oR$)$^{-1}$ | $3.97 \times 10^5$ |

Viscosity of Steam, $\mu$, lb. sec./in.$^2$ at T = 800°F — $3.52 \times 10^{-9}$

Desirable bearing gas flow, lb/sec. — $\leq$ .183 or 5 percent total turbine flow

The journal bearing loads are as follows:




$$\Sigma M_1 = 0$$

$$\therefore W_2 = \left(\frac{6.783}{10.875}\right) 154.5 = 96.37 \text{ lbs.}$$

$$\Sigma M_2 = 0$$

$$W_1 = \left(\frac{4.092}{10.875}\right) 154.5 = 58.2 \text{ lbs.}$$

Using the criterion presented on page 12 of MTI-65TR5-II, Vol. 2, to select single- or double-plane admission, which states that at best only 25 percent of the supply pressure acting over the bearing surface is available to carry load.

For the compressor end:

$$\frac{W_1}{(P_s - P_a)\, LD} = \frac{58.2}{(350-30)(2)(4)} = .0275$$

For the turbine end:

$$\frac{W_2}{(P_s-P_a)\ LD} = \frac{96.37}{(350-30)(5.375)(2.6875)} = .0208$$

For both bearings, the criterion for single-plane admission is satisfied. However, it states in the preliminary specifications for the forced-draft-blower that the complete blower must be capable of sustaining shock and vibration loads of intensity and frequency normally encountered in service. This being the case, the greater stiffness offered by the double-plane admission will be considered.

For the off-design condition of $P_s = 200$ psia

$$\frac{W_1^1}{(P_s-P_a)\ LD} = \frac{58.2}{(200-30)(2)(4)} = .0517$$

$$\frac{W_1^2}{(P_s-P_a)\ LD} = \frac{96.37}{(200-30)(5.375)(2.6875)} = .0391$$

Both bearings satisfy the criterion for single-or double-plane admission in the off-design supply pressure condition.

When the radial clearance of the journal bearing is to be considered, four items must be taken into consideration. These are:

1. Bearing film stiffness (a linear function of the clearance).
2. Radial growth due to centrifugal force.
3. Radial growth due to thermal expansion.
4. Flow.

The radial growth due to the centrifugal force can be determined as:

$$\delta_c = \frac{8.04}{E} R^3$$

$$\delta_c = +0.00048 \text{ in.}$$

This growth is that of the rotating bearing shell and is an increasing value with respect to speed. The radial growth due to temperature has been calculated

to be:

$$\delta_t = -0.00074 \text{ in.}$$

In this calculation, the rotating bearing is cooler than the stationary shaft — causing a decrease in the bearing radial clearances.

The total change in the radial clearance would be:

$$\delta = \delta_c + \delta_t$$

$$\delta = +0.00048 - 0.00074$$

$$\delta = -0.00026$$

This would be the total change in radial clearance from a cold start to running at design temperature.

It is felt that the total change in film thickness due to radial growth from the centrifugal force and thermal conditions are in the proper direction. The film thickness presented in this text are at design condition; machining allowances must be made for this change in growth at startup.

The suggested range in clearance to radius ratio (C/R) is given on page 13 of MTI-65TR5 - II, Vol. 2, as $0.5 \times 10^{-3}$ to $1 \times 10^{-3}$. The selected C/R in this case is to be $0.75 \times 10^{-3}$ at operating speed. This should allow for sufficient stiffness, moderate flow and reasonable manufacturing clearances at both the cold start condition and design condition. For these journal bearings the operating clearances would be:

for the compressor end (Station 1):

$$C_1 = (0.75 \times 10^{-3})\ (4/2) = 1.5 \times 10^{-3} \text{ inch,}$$

for the turbine end (Station 2):

$$C_2 = (0.75 \times 10^{-3})\ \left(\frac{5.375}{2}\right) = 2.02 \times 10^{-3} \text{ inch.}$$

as the cold start condition:

$$C_1 = 1.5 \times 10^{-3} - 0.26 \times 10^{-2}$$
$$= 1.24 \times 10^{-3} \text{ inch.}$$
$$C_2 = 2.02 \times 10^{-3} - 0.26 \times 10^{-3}$$
$$= 1.76 \times 10^{-3} \text{ inch.}$$

The consideration of flow with respect to clearance will be discussed after the selection of the other bearing parameters.

The design charts do not present data for double-plane admission for L/D = 0.5. However, the results obtained from the design charts for L/D = 1, 1.5, and 2 are obtained at a pressure ratio $P_s/P_a$ = 350/30 = 11.7 and extrapolated to L/D = 0.5.

These data and extrapolations are shown in Fig. IV-3. The data shown are obtained at the optimum dimensionless stiffness. By definition, the optimum dimensionless stiffness is the maximum value of the dimensionless stiffness for the entire range of restrictor coefficient. The values of the bearing parameters obtained from the extrapolation are:

$$\bar{K} = 0.7$$
$$\Lambda_s \xi = 0.5$$
$$\bar{Q} = 0.3$$

The dimensionless stiffness, $\bar{K}$, is defined on page 9, MTI-65TR5 - II, Vol. 2 as

$$\bar{K} = \frac{1 + \delta^2}{1 + \frac{2}{3}\delta^2} \frac{CK}{(P_s - P_a)\, LD}$$

As explained in the text, in the case of inherent compensation where S >> 1

$$\bar{K} = \frac{3}{2} \frac{CK}{(P_s - P_a)\, LD}$$

from which

$$K = \frac{2}{3} \frac{(P_s - P_a)}{C} LD\bar{K}$$

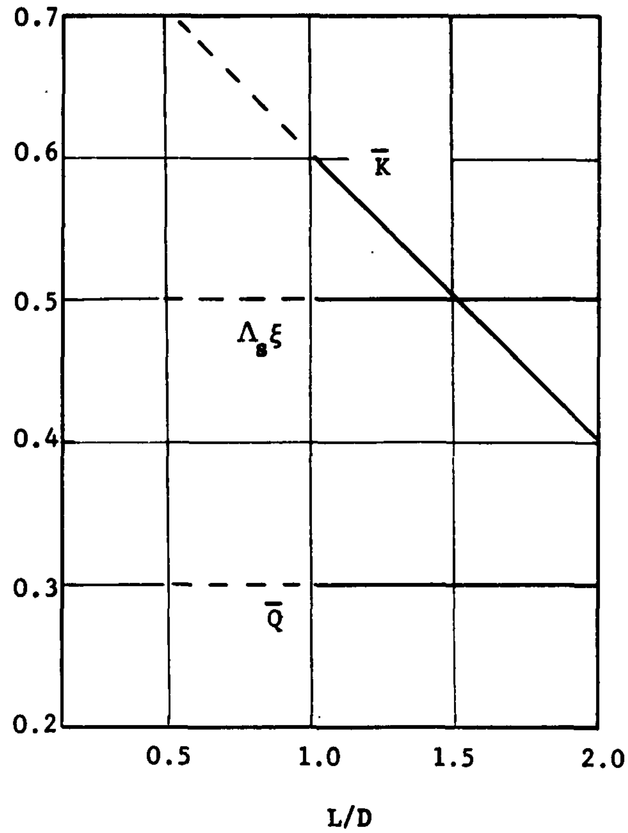



Figure IV-3

For the compressor-end journal bearing

$$K_1 = \frac{2}{3} \frac{(350-30)}{(1.5 \times 10^{-3}} \; 2.4 \; . \; 0.7$$

$$= 7.97 \times 10^5 \text{ lb/in.}$$

For the turbine-end journal bearing

$$K_2 = \frac{2}{3} \frac{(350-30}{(2.02 \times 10^{-3}} \; 5.375 \; . \; 2.6875 \; . \; 0.7$$

$$= 10.68 \times 10^5 \text{ lb/in.}$$

Using these bearing film stiffnesses, critical speed calculations were made that showed the first and second critical occurring at 9000 and 21,000 RPM respectively. This satisfies the criterion on hydrodynamic instability in that the lowest critical frequency must be greater than 0.6 times operating speed.

The eccentricity ratio is defined on page 22, MTI-65TR5 - II, Vol. 2, as

$$\epsilon = \frac{W}{CK}$$

For the compressor-end journal bearing

$$\epsilon_1 = \frac{58.2}{(1.5 \times 10^{-3})(7.97 \times 10^5)}$$

$$\epsilon_1 = .049$$

For the turbine-end journal bearing

$$\epsilon_2 = \frac{96.37}{(2.02 \times 10^{-3})(10.68 \times 10^5)}$$

$$\epsilon_2 = .045$$

The eccentricity ratios of both journal bearings satisfy the criterion for preventing lockup, i.e., in order to prevent lockup, the eccentricity ratio should be less than 0.4.

The dimensionless flow is defined on page 17, MTI-65TR5 - II, Vol. 2, as

$$\overline{Q} = \frac{6\mu RT}{\pi C^3 P_s^{\,2}} \xi Q$$

from which

$$Q = \frac{\pi C^3 P_s^{\,2} \overline{Q}}{6\mu RT\xi}$$

For the compressor-end journal bearing

$$Q_1 = \frac{\pi(1.5 \times 10^{-3})^3 \ (350)^2 \ (0.3)}{6(3.52 \times 10^{-9})(3.97 \times 10^5)(1260)\frac{1}{4}}$$

$$Q_1 = 1.47 \times 10^{-4} \text{ lb.sec/in.}$$

$$Q_1 = 5.7 \times 10^{-2} \text{ lb/sec.}$$

For the turbine-end journal bearing

$$Q_2 = \frac{\pi(2.02 \times 10^{-3}) \ (350)^2 \ (0.3)}{6(3.52 \times 10^{-9})(3.97 \times 10^5)(1260)(\frac{1}{4})}$$

$$Q_2 = 3.6 \times 10^{-4} \text{ lb.sec/in.}$$

$$Q_2 = 14 \times 10^{-2} \text{ lb/sec.}$$

These journal bearings represent 5.38 percent total turbine flow. The selection of the number, N, and size, d, of the feeder holes is now in order. From page 19, MTI-65TR5 - II, in the case of inherent compensation

$$Nd = \frac{P_s C^2}{6\mu\sqrt{RT}} \Lambda_s$$

For the compressor-end journal bearing

$$(Nd)_1 = \frac{(350)(1.5 \times 10^{-3})^2 (2)}{6(3.52 \times 10^{-9})(3.97 \times 10^5)(1.260 \times 10^3)} \frac{1}{2}$$

$$(Nd)_2 = 3.345$$

For the turbine-end journal bearing

$$(Nd)_2 = \left[\frac{*350)(2.02 \times 10^{-3})^2 (2)}{6(3.52 \times 10^{-9})(3.97 \times 10^{5})(1.26 \times 10^{3})}\right]^{\frac{1}{2}}$$

$$(Nd)_2 = 4.5$$

Figure V, MTI-65TR5 - II, Vol. 2, presents the criterion for the minimum number versus the diameter of the feeder holes. While in Part II, Vol. 2, the technique used is to evaluate $\frac{d}{\xi D}$ and solve for $N\xi)_{min}$, the technique here will be reversed, i.e., evaluate $N\xi)$, and solve for $\frac{d}{\xi D}_{min}$. The motivation for this is in selecting a large number of orifices to prevent the possibility of lockup.

Therefore, let the number of orifices be 12 per feeder plane for a total of 24 per journal bearing.

$$N\xi = 24\ (\tfrac{1}{4}) = 6$$

From Figure V, MTI-65TR5 - II, Vol. 2,

$$\left.\frac{d}{\xi D}\right)_{min} = 7 \times 10^{-2}$$

For the compressor-end journal bearing

$$d_1)_{min.} = \tfrac{1}{4}\ (4)(7 \times 10^{-2})$$

$$= .07 \text{ inch}$$

For the turbine-end journal bearing

$$d_2)_{min.} = \tfrac{1}{4}\ (5.375)(7 \times 10^{-2})$$

$$= .094 \text{ inch}$$

The actual value of d

$$d_1 = 3.345/24$$

$$= .1395 \text{ inch}$$

$$d_2 = 4.5/24$$
$$= .1875 \text{ inch}$$

At this point, there is sufficient information available on the design to approximate the entrance effect pressure drop.

The pressure downstream of the inherent compensation is found by the procedure outlined on page 18, MTI-65TR5 - II, Vol. 2,

$$\bar{Q}_o = \frac{\bar{Q}}{\Lambda_s \xi}$$
$$= \frac{0.3}{0.5} = 0.6$$

Using Figure IV, MTI-65TR5 - II, Vol. 2, at a value of $\bar{Q}_o = 0.6$

$$\frac{P_c}{P_s} = 0.717$$

or $P_c = (0.717)(350)$

$$P_c = 251 \text{ psia}$$

This pressure drop from supply pressure to the downstream orifice pressure is shown on the Mollier diagram (Figure IV-4) as an isentropic expansion from point (A) to (B).

The entrance effect pressure drop, also assumed to be an isentropic expansion is calculated as:

$$\Delta p = \frac{\rho V^2}{2g} = \frac{V^2}{2\nu g}$$

Now $Q = AV/\nu$

$$V = Q\nu/A$$

$$\therefore \Delta p = \frac{\nu}{2g} \left(\frac{Q}{A}\right)^2$$

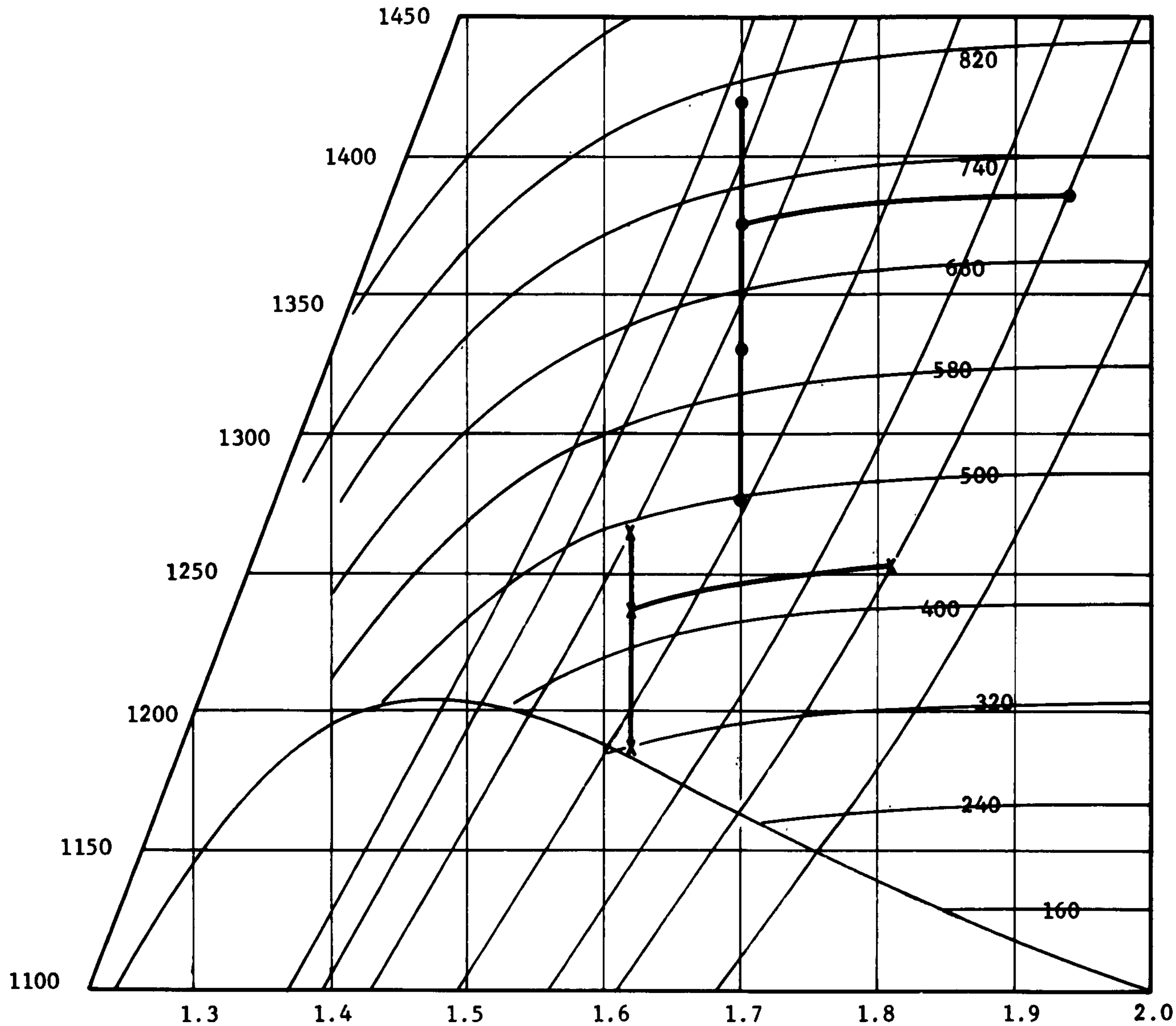



Fig. IV-4 Thermo-Dynamic Chart Showing Bearing Steam Conditions

where A = the annular surface surrounding the feeder holes = N $\pi$dC

$$\Delta p = \frac{\nu}{2g} \left(\frac{Q}{N\pi dC}\right)^2$$

At P = 251 psia

T = 710 F (from the Mollier diagram)

$\nu$ = 2.707 $ft^3/lb$ (from steam tables)

= $4.67 \times 10^3$ $in^3/lb$.

For the compressor-end journal bearing

$$(\Delta p)_1 = \frac{4.67 \times 10^3}{2(386.07)} \left(\frac{5.7 \times 10^{-2}}{24\pi(.1395)(1.5 \times 10^{-3})}\right)^2$$

$$(\Delta p)_1 = 78.5 \text{ lb/in.}^2 \text{ abs.}$$

$$(P_c - \Delta p)_1 = 251 - 78.5$$

$$= 173 \text{ lb/in.}^2 \text{ abs.}$$

This is shown as point (C) on the Mollier diagram, Fig. IV-4.

For the turbine-end journal bearing

$$(\Delta p)_2 = \frac{(4.67 \times 10^3)}{2(386.07)} \left(\frac{14 \times 10^{-2}}{24\pi(.1875)(2.02 \times 10^{-3})}\right)^2$$

$$(\Delta p)_2 = 145.5 \text{ lb/in.}^2 \text{ abs.}$$

$$(P_c - \Delta p)_2 = 251 - 146$$

$$= 105 \text{ lb/in.}^2 \text{ abs.}$$

This is shown as point (D) on the Mollier diagram.

The pressure levels as they exist in the bearing are now represented on the Mollier diagram. For the compressor-end journal bearing, the pressure drop is shown as an isentropic drop through the annulus surrounding the feeder hole

(points A to B), then as an isentropic pressure drop due to the entrance effect (points B to C), recovery to point B and then an isothermal drop from point B to E. For the turbine-end journal bearing, the pressure level changes are shown as points A to B to D to B to E. It should be repeated here that this is an approximate method and is considered to be conservative.

The following is a summary of the journal bearing dimensions and the design conditions:

| | Compressor end | Turbine end |
|---|---|---|
| Diameter, D, in. | 4 | 5.375 |
| Length, L, in. | 2 | 2.6875 |
| Length between admission planes, L., in. | 1 | 1.342 |
| Radial clearance, C, in. | 0.0015 | .00202 |
| No. of Admission planes | 2 | 2 |
| No. of feeder holes per plane | 12 | 12 |
| Feeder hole diameter, d, in. | .1395 | .1875 |
| Supply pressure, $P_s$, psia | 350 | 350 |
| Supply temperature, T, °F | 800 | 800 |
| Flow Q, lb/sec. | .057 | .14 |
| Radial stiffness, K, lb/in. | $7.97 \times 10^5$ | $10.68 \times 10^5$ |
| Operating eccentricity ratio | .049 | .045 |
| Bearing load, L, lbs. | 58.2 | 96.4 |

The "off-design" conditions for the journal bearing require that the bearings operate with 200 psia saturated steam. This condition arises at times while the ship is docked or while maneuvering at sea. Since the bearings will not operate with a two-phase lubricant, it will be necessary to provide a superheater as auxiliary equipment which will superheat the steam to a temperature level high enough so that the pressure drop at the annulus surrounding the entrance hole and pressure drop due to the entrance effect will not be enough to cause any saturated condition to exist within the bearing.

The calculations for the bearing were repeated under the 200 psia supply pressure condition. The summary of the bearing under these conditions are as follows:

| | Compressor end | Turbine end |
|---|---|---|
| Supply pressure, $P_s$, psia | 200 | 200 psia |
| Supply temperature, T, °F (requiring at least 110 F superheat to be supplied from an auxiliary device) | 490 | 490 |
| Flow, Q, lb/sec. | .0355 | .087 |
| Radial stiffness, K, lb/in. | $4.11 \cdot 10^5$ | $5.2 \cdot 10^5$ |
| Operating eccentricity | .095 | .092 |

The pressure levels obtained in the bearings are shown on the Mollier diagram with the prime notation.

It should be repeated that this is an approximate method based on the following assumptions:

1. The pressure drop of the annular control area is isentropic
2. The pressure drop due to entrance effects is isentropic
3. Adiabatic conditions exist between the steam and the bearing surfaces.

The thrust bearing for the blower is to support a unidirectional thrust load acting toward the compressor end. However, it is desired that the bearing used be a double-acting type in which the normally unloaded thrust bearing would act as a positioner, i.e., if the thrust load were to become zero, the gas impinging on the runner would cause the runner to move away from the bearing surface unrestrained. The action of the positioner is to restrain this motion — nothing more.

The specifications with respect to the lubricant and the ambient conditions are the same for the thrust bearing as for the journal bearing. The unique specifications are:

| | |
|---|---|
| Thrust load | 2500 lbs. |
| Proposed outside radius, $R_1$, in. | |
| loaded bearing | 3.25 |
| unloaded bearing | 3.25 |
| Proposed inside radius, $R_2$, in. | |
| loaded bearing | 1.00 |
| unloaded bearing | 2.5 |
| Radius Ratio | |
| loaded bearing | 3.25 |
| unloaded bearing | 1.45 |

(The design curves for $R_1/R_2 = 3.0$ will be used for the loaded bearing and for $R_1/R_2 = 1.25$ for the unloaded bearing)

The loaded bearing will be chosen to be an optimum bearing, i.e., the maximum value of dimensionless stiffness over the range of restrictor coefficient. Using Figs 28, 34, and 38, as $P_s/P_a = 11.7$ one obtains

$$\bar{K} = 0.42 \text{ (optimum)}$$
$$\Lambda_s = 0.7$$
$$\bar{W} = 0.27$$
$$\bar{Q} = 0.46$$

Therefore, the load

$$W = \pi(R_1^2 - R_2^2)(P_s - P_a)\bar{W}$$
$$W = \pi(3.25^2 - 1^2)(350-30)(0.27)$$
$$W = 2595 \text{ lb.}$$

This bearing would support the 2500 lb. due to the external thrust load plus 95 lb. applied by the unloaded bearing.

For the stiffness on the loaded bearing

$$CK = \pi(R_1^2 - R_2^2)(P_s - P_a)\bar{K}$$
$$= \pi(3.25^2 - 1.^2)(350-30)(.42)$$
$$CK = 4040. \text{ lb.}$$

At the clearance, C, = 0.0015

$$K = 2.695 \cdot 10^6 \text{ lb/in.}$$

For the flow on the unloaded bearing

$$Q/C^3 = \frac{\pi P_s^2 \bar{Q}}{6\mu RT}$$

$$\frac{Q}{C^3} = \frac{\pi(3.5\cdot 10^2)^2(.46)}{6(3.52\cdot 10^{-9})(3.97\cdot 10^5)(800 \times 460)}$$

$$Q/C^3 = 1.675\cdot 10^4$$

at C = .0015

$Q = 5.65\cdot 10^{-5}$ lb. sec/in.

$Q = .02185$ lb/sec.

Using the equations and the procedure outlined on page 30, MTI-65TR5 - II, Vol. 2, for determining the number and size of the feeder holes:

$$Nd = \frac{P_s C^2 \Lambda_s}{6\mu \sqrt{RT}}$$

$$Nd = \frac{(350)(1.5\cdot 10^{-3})^2(0.7)}{6(3.52\cdot 10^{-9})\sqrt{(3.97\cdot 10^{+5})(1.26\cdot 10^3)}}$$

$$Nd = 1.17$$

$$\xi = \tfrac{1}{2}\, \ell n \left(\frac{R_1}{R_2}\right)$$

$$= \tfrac{1}{2}\, \ell n \left(\frac{3.25}{1}\right)$$

$$= .59$$

$$D = 2\sqrt{(3.25)(1.0)}$$

$$D = 3.6$$

Selecting

$$N = 8$$

$$d = 1.17/8$$

$$= .146 \text{ inch}$$

$$\frac{d}{\xi D} = \frac{.146}{(.59)(3.6)} = .0686$$

From Figure V, MTI-65TR5 - II, Vol. 2,

$$(N\xi)_{min} = 4.6$$

$$N_{min} = 4.6/.59 = 7.8$$

The criterion stated on page 30, MTI-65TR5 - II, Vol. 2, is satisfied.

The calculation for the pressure downstream of the inherent compensation surface is done as explained on page 30 for the loaded thrust bearing.

$$Q_o = \frac{\bar{Q}}{\Lambda_s} = \frac{0.46}{(0.7)}$$

$$Q_o = .657$$

From Figure IV, MTI-65TR5 - II, Vol. 2,

$$P_c/P_s = .628$$

from which

$$P_c = (.628)(350)$$

$$= 220 \text{ psia}$$

It is now possible to approximate the entrance effect pressure drop from

$$\Delta p = \frac{\nu}{2g} \left(\frac{Q}{N\pi dC}\right)^2$$

at $Pc = 220$ psia

$T = 675$ F(from the Mollier Chart)

$\nu = 2.992$ ft.$^3$/lb. (from Steam Tables)

$= 5.16 \cdot 10^3$ in.$^3$/lb.

$$\Delta p = \frac{(5.16 \cdot 10^3)}{(2)(3.8607 \cdot 10^2)} \left[\frac{2.185 \cdot 10^{-2}}{8\pi(.146)(1.5 \cdot 10^{-3})}\right]^2$$

$$\Delta p = 105.5 \text{ psia}$$

The minimum pressure obtained due to entrance effects is then

$$(P_c - \Delta p) = 220-105.5 = 114 \text{ psia}$$

This pressure is above the saturation line.

For the unloaded side of the thrust bearing, the load to be applied by this bearing is 95 lbs. Therefore,

$$\overline{W} = \frac{W}{\pi(R_1^2 - R_2^2)(P_s - P_a)}$$

$$= \frac{95}{\pi(3.25^2 - 2.5^2)(350-30)}$$

$$= .0219$$

Using the design charts for $R_1/R_2 = 1.25$ at a pressure ratio of 12, one obtains

$$\Lambda_s = .1 \quad \text{(Figure 27)}$$

$$\overline{Q} = 7 \cdot 10^{-2} \quad \text{(Figure 31)}$$

$$\overline{K} = 5.9 \cdot 10^{-2} \quad \text{(Figure 35)}$$

Converting the dimensionless quantities, one obtains for the stiffness

$$K = \frac{\pi(R_1^2 - R_2^2)(P_s - P_a)\ \overline{K}}{C}$$

$$K = \frac{\pi(3.25^2 - 2.5^2)(350-30)(5.9 \cdot 10^{-2})}{C}$$

$$K = 256/C$$

for the flow

$$Q = \left(\frac{\pi 2_s^{\ 2}}{6\mu RT}\ \overline{Q}\right) C^3$$

$$Q = \frac{\pi(3.5 \cdot 10^2)(7 \cdot 10^{-2})\ C^3}{6(3.52 \cdot 10^{-9})(3.97 \cdot 10^5)(1.260 \cdot 10^3)}$$

$$Q = 2.55 \cdot 10^3 \ C^3$$

With the film thickness on the unloaded bearing being chosen as 0.002 inches as normal operating conditions,

$$K = 128000 \text{ lb/in.}$$

$$Q = 2.040 \cdot 10^{-5} \text{ lb. sec/in.}$$

$$Q = 7.89 \cdot 10^{-3} \text{ lb/sec.}$$

The criterion outlined on page 30, MTI-65TR5 - II, Vol. 2, for determining the number and size of feeder holes required serves to validate the analysis. In order for the design curves to accurately predict the load, etc., this criterion, with respect to the minimum number of feeder holes, must be satisfied. However, in the case of the unloaded side of the thrust bearing, the requirement to carry a load is not stringent enough to permit no deviation from the design charts. The unloaded face of the thrust bearing is to act as a positioner only. Therefore, the selection of the size and number of the feeder holes will not be applied in this case. The result of this will be that the calculated load will be somewhat in error.

From the definition of the restrictor coefficient

$$Nd = \frac{P_s \ C^2 \ \Lambda_s}{6\mu \sqrt{RT}}$$

$$Nd = \frac{(350)(2.10^{-3})^2(.1)}{6(3.52 \cdot 10^{-9}) \sqrt{(3.97 \cdot 10^{5})(1.26 \cdot 10^{3})}}$$

$$Nd = .296$$

Selecting N = 8, i.e., a feeder hole at every 45 degrees,

$$d = .296/8$$

$$d = .037 \text{ inch}$$

For the entrance effect pressure drop

$$\overline{Q}_o = \overline{Q}/\Lambda_s$$

$$= (7\cdot 10^{-2}) / \ (.1)$$

$$\bar{Q}_o = .7$$

Using Figure IV, MTI-65TR5 - II, Vol. 2, one finds that the flow is choked. The downstream pressure, $P_c$, is therefore

$$P_c = P_s\,(0.528)$$

$$= 350\,(0.528)$$

$$= 185 \text{ psia}$$

For the entrance effect approximation

$$\Delta p = \frac{\nu}{2g}\left(\frac{Q}{\pi N d c}\right)^2$$

at $P_c = 185$ psia

$T = 640$ F(From the Mollier Chart)

$\nu = 3.425$ ft.$^3$/lb (from the Steam Table)

$= 5.91\cdot 10^3$ in.$^3$/lb.

$$\therefore \Delta p = \frac{(5.91\cdot 10^3)}{2(386.07)}\left[\frac{7.89\cdot 10^{-3}}{8\pi(.037)(2.10^{-3})}\right]$$

$$\Delta p = 133 \text{ psia}$$

The lowest pressure attained due to entrance effect is

$$(P_c - \Delta p) = 185-133$$

$$= 52 \text{ psia}$$

It can be seen that on the Mollier diagram this value is also above the saturation line. The bearing will not experience phase change of the lubricant within the film.

The following is a summary of the thrust bearing dimensions and design conditions:

| | Loaded Side | | Unloaded Side |
|---|---|---|---|
| Outer Diameter, $2R_1$, in. | 6.50 | | 6.50 |
| Inner Diameter, $2R_2$, in. | 2.0 | | 5.0 |
| Design Load, L, lbs. | 2600 | | 100 |
| Total Applied load, $L_t$, lbs. | | 2500 | |
| Design Film Thickness, C, in. | .0015 | | .002 |
| Number of feeder holes, N. | 8 | | 8 |
| Feeder hole diameter, d. | .146 | | .037 |
| Type of compensation | | inherent | |
| Diameter of feeder hole circle, $R_1R_2$, in. | 3.6 | | 5.71 |
| Design film stiffness, K, lb/in. | 2,695,000. | | 128,000 |
| Design flow, Q, lb/sec. | .02185 | | .00789 |
| Supply pressure, $P_s$, $lb/in.^2$ | 350 | | |

The off-design conditions for the thrust bearing are the same as required for the journal bearing, i.e., the steam conditions can drop as low as 200 psia saturated. As is seen in the calculations for the journal bearing, it will be necessary to provide a superheater as auxiliary equipment. This is to maintain enough superheat in the steam to prevent the formation of the dual phase of the lubricant. In the loaded thrust bearing, this is still the case. However, in the unloaded thrust bearing the loss of load-carrying ability is not stringent since it acts as a positioner only. Formation of the second phase of steam could possibly cause errosion of the bearing surfaces if the unit were to operate under these conditions for an extended time period. This is considered here to be a secondary problem. Main emphasis will be directed towards preventing the loss of load carrying ability in the loaded thrust bearing.

The calculations for the thrust bearing were repeated under the 200 psia supply pressure condition. Just the summary of the calculations performed are shown here.

| | Loaded Thrust Face | Unloaded Thrust Face |
|---|---|---|
| Supply pressure, $P_s$, psia | 200 | 200 |
| Supply temperature, T, $^oF$ | 490 | 490 |
| Film thickness, C, in. | .00046 | .003 |
| Load, L, lbs. | 2500 | 20 |
| Flow, Q, lb/sec. | $8.607 \cdot 10^{-4}$ | $2.28 \cdot 10^{-3}$ |
| Axial stiffness, K, lb/in. | $5.77 \cdot 10^5$ | $1.92 \cdot 10^4$ |

| | Loaded Thrust Face | Unloaded Thrust Face |
|---|---|---|
| Pressure downstream of inherent compensation, $P_c$, psia | 197 | 105.5 |
| Pressure due to entrance effect, $\Delta p$, psia | 1.5 | 6.65 |
| Lowest pressure level developed at the isentropic pressure drop, $(P_c-\Delta p)$ psia | 195.5 | 98.9 |

One will note that under the conditions specified above that the film on the loaded thrust bearing will be only 0.0005 in. For this diameter bearing, a film thickness of this level is extremely small. There are several ways of overcoming this. These are:

1. Increase the number or size of feeder holes.
2. Increase the total area of the bearing.
3. Change the design to have a double set of feeder holes at two different diameters.
4. Restrict the bearing design to operate for the minimal steam conditions.

The first method provides the simplest cure; the effect would be that as the design conditions of $P_s$ = 350 psia and T = 700 F, the film thickness would be greater than the 0.0015 specified. This would also result in higher flows at the design condition. For a comparison, the following table is provided in which the film thickness is specified to be 0.00075 in. at the off-design condition of $P_s$ = 200 psia.

Loaded Thrust Bearing

| | | |
|---|---|---|
| Supply pressure, $P_s$, psia | 200 | 350 |
| Supply temperature, T, °F | 490 | 800 |
| Film thickness, C, in. | 0.00075 | .00245 |
| Number of feeder holes, N. | 8 | 8 |
| Feeder hole diameter | .146 | .389 |
| Load, L, lbs. | 2500 | 2500 |
| Flow, Q, lb/sec. | .0219 | .0952 |

The effect of the change in film thickness is readily apparent, especially with respect to the flow. The increase in the total area of the bearing would not cause an increase in flow; in fact, there would be a decrease.

Heat generated in the hydrostatic bearings is carried away by the lubricant flow. Thus, the clearances are controlled by the thermal environment of the bearing rather than the bearing losses.

It is felt that the most effective means of dealing with the wide variation in supply conditions and minimal changes in the film would be the third choice. i.e., the use of a two-row set of orifices. However, this approach is beyond the scope of this report. The fourth choice would be to design the bearings for the 200 psia condition, either regulating the 350 psia steam down to this value, when 350 psia is the available pressure, or designing the bearing to operate at whatever pressure is available.

Differences in back pressure on the bearings at a given steam feed pressure will change the load carrying ability of the bearing. This affect is small, however, for the feed pressure considered here and for fluctuations of only a few psi. Off design calculations will be necessary if any significant back pressure changes are to be encountered in practice. This, again, is beyond the scope of this report.

The above three considerations and choices are better made the subject of a test program in development, and so are beyond the scope of this report.

## CONCLUSION

This forced draft blower design presents many attractive features. The unit has good performance, it is simple and rugged, and it is small and lightweight. This means a great saving to the user in the form of low initial cost and low maintenance, operating, and installation expenses. The use of gas bearings coupled with a high performance blower design, allows the significant improvement possible with the proposed turbo-blower.

The steps indicated in the Bearing Design section clearly show the considerations necessary to make a successful gas-bearing application. The same types of system considerations are necessary for either hydrostatic or hydrodynamic bearings although the detailed bearing designs are very different. The general application considerations given here together with the examples given in MTI-65TR5 - II, Vol. 2, and the detailed bearing design illustrated here should be sufficient to instruct the reader on methods of evaluating gas bearing applications.

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13. ABSTRACT

The application of gas bearings to an actual machine design — The Forced-Draft Blower — is illustrated. The design represents a machine which takes full advantage of the potential gained by using gas bearings. This bearing application study relates the bearing design to the curves presented in Part II of this report. Emphasis is placed upon the methods of fixing the design. The other important system features are checked to determine the adequacy of the design — stress, temperatures, steam conditioning, and rotor response.

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